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Computational Fluid Dynamics Analysis of Mixed Convection Heat Transfer and Fluid Flow in a Lid-driven Square Cavity Subjected to Different Heating Conditions

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Abstract-

The present study investigates mixed convective and fluid flow characteristics in a lid-driven enclosure filled with air and its walls subjected to various heating conditions. The vertical (left and right) walls of the enclosure are cooled (T_c), and the bottom wall is heated to (T_h) while the horizontal lid-driven upper wall is subjected to sinusoidal heating. The dimensionless governing equations (continuity, momentum, and energy transport) were implemented in COMSOL Multiphysics 5.4 software. The influences of Grashof number ($10^3 \leq Gr \leq 10^5$) and Reynold number in the interval of $1 \leq Re \leq 100$ on the average Nusselt number ($\overline{Nu}$) for all walls of the cavity was examined. Furthermore, the results presented in the form of isotherms, streamlines, and the local and average Nusselt numbers in the enclosure for $Re = 100$ and Gr in the range of $10^3 \leq Gr \leq 10^5$. The results indicated the highest and lowest average rate of heat transfer at the bottom and top walls of the cavity respectively. The top wall region presented a higher velocity as confirmed by the velocity contour plots.

Keywords: mixed convection, CFD, differentially heated square, Lid-driven

1. Introduction

Many researchers have undertaken studies on mixed convection in lid-driven enclosures using both numerical and experimental approaches because of its huge benefits to the engineering field. It found applications in heat exchangers, cooling of electronic devices, space heating, nuclear as well as in chemical reactors [1-3]. Torrance et al. [4] examined fluid flow in a lid-driven enclosure and concluded that aspect ratio increment resulted in a corresponding increase in secondary circulation in the lower section of the cavity. In a numerical investigation of mixed convection in a driven enclosure, Iwatsu et al. [5] submitted that high Richardson number causes stagnation of the fluid in the middle and lower regions of the cavity. Ducasse and Sibanda [6] simulated a similar atmosphere with sinusoidal moving lid and controlled heat flow through the sidewalls. One of the conclusions drawn was that, for the case of the heated sidewalls, both of the average and the local Nusselt numbers are lower compared to the walls subjected to heating.

Mixed convection was analysed by Coasta and Raimundo [7] with a cylinder rotating in a square cavity; with differential heating conditions imposed on the walls of the enclosure. It was opined that the processes of heat transfer are a function of the size of the enclosure cylinder. In



another numerical study of this same concept, Abraham and Varghese [8] insulated the vertical walls of the cavity and subjected the horizontal upper and lower walls to the highest and lowest constant temperatures respectively. In one of the scenarios considered, the horizontal walls moved in the same direction while in the second case, the movement was in opposite directions. It was submitted that Richardson number is the parameter that dictates whether heat transfer is purely convective, conductive, or mixed convection.

Shuja et al. [9] investigated the impacts of a range of aspect ratios, exit port cavity, and a rectangular heat generating body placed inside a square cavity on the heat transfer characteristics in the enclosure. It was concluded that beyond exit ports location number of 13, buoyancy driven flow is enhanced and, the irreversibility to heat transfer ratio was found to reduce for low exit port location numbers. Doghmi et al. [10] analyzed a 3-D mixed convection in a ventilated square cavity with a block located in it; one of the major conclusions made was that for a specified Richardson number, the average Nusselt number (Nu) decrease with decreasing the Reynolds number.

Unlike Abraham and Varghese [8], Nosonov and Sheremet [11] used a rectangular cavity with a heater situated inside the cavity in their study of natural convection. Heat transfer characteristics were analyzed for Richardson numbers at a Reynolds number of 100 using air as the working fluid. It was found that increased Richardson number resulted in increased intensity of internal convective vortex and that heater location modifies heat transfer rate. Abdulsahib and Al-Farhany [12] experimentally investigated mixed convective heat transfer in a porous nanofluids-filled enclosure with a circular cylinder rotating inside of it. One of the conclusions made was that low permeability of the medium affects temperature distribution and vortices production depends on the angular velocity and direction of cylinder rotation.

Oztop and Dagtekin [13], in another study, did an in-depth analysis of steady state mixed convection in a square enclosure whose vertical sides are driven by lids. In their study, the horizontal walls adiabatically while subjecting the vertical walls to steady temperatures. The conclusion drawn was similar to those of earlier researchers as it affects the dependence of heat transfer characteristics of the cavity on Richardson number and direction of lid-driven walls. Omari [14] in his study of different aspect ratios of a lid-driven cavity subjected to laminar density invariant fluid flow inferred that pressure reduction is dependent on the aspect ratio and Reynolds number which is evident in the structure of the primary vortex.

This study seeks to use computational fluid dynamics to analyze mixed convection heat transfer and fluid flow in a lid-driven square cavity subjected to different heating conditions.

2. Methodology

In the current section, the physics of the present investigation, the boundary conditions applied, the normalized continuity and transport equations are discussed along with the parameters that served as input into COMSOL Multiphysics 5.4 software.

2.1 Problem description

Figure 1a is a square cavity having its boundaries exposed to various thermal conditions. The left and right cold boundaries are subjected to a dimensionless temperature $\varphi_c = 0$, the bottom hot boundary is kept at a temperature $\varphi_h = 1$ and the top horizontal boundary is held at a

sinusoidal temperature $\varphi_{h1} = \sin(\pi X)$ and moving with a velocity $U = U_0$ in the positive x -direction. The working fluid is taken to be viscous, density invariant, Newtonian, and obeys the Boussinesq model of approximation. Furthermore, the flow is assumed to be steady with negligible radiation effects. Figure 1b presents the mesh dimension used for the numerical studies.

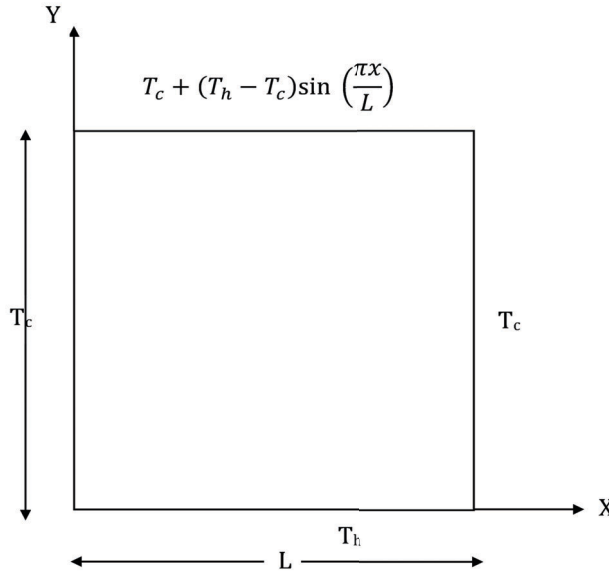


Figure 1a: Physical model

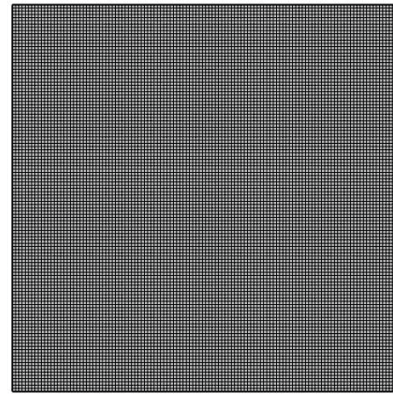


Figure 1b: Mesh distribution

2.2 The governing equations

The normalized equations governing the present study are given as follows [13]:

Continuity:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (1)$$

X- Momentum:

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{Re} \left[\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right] \quad (2)$$

Y-: Momentum:

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{Re} \left[\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right] + \frac{Gr}{Re^2} \varphi \quad (3)$$

Energy:

$$U \frac{\partial \varphi}{\partial X} + V \frac{\partial \varphi}{\partial Y} = \frac{1}{Re \cdot Pr} \left[\frac{\partial^2 \varphi}{\partial X^2} + \frac{\partial^2 \varphi}{\partial Y^2} \right] \quad (4)$$

2.3 Normalization parameters

The normalization parameters used to transform the relevant dimensional equations into normalized equations are given by[13]:

$$X = \frac{x}{L}, Y = \frac{y}{L}, U = \frac{u}{U_0}, V = \frac{v}{U_0}, \phi = \frac{T-T_c}{T_h-T_c}, P = \frac{p}{\rho U_0^2}, Pr = \frac{\nu}{\alpha} \quad (5)$$

Equation (6) gives the local Nusselt number on the walls as shown below:

$$Nu = k \left. \frac{\partial \phi}{\partial X} \right|_{x=L} \quad (6)$$

The average Nusselt number on the solid boundaries of the configuration is given by

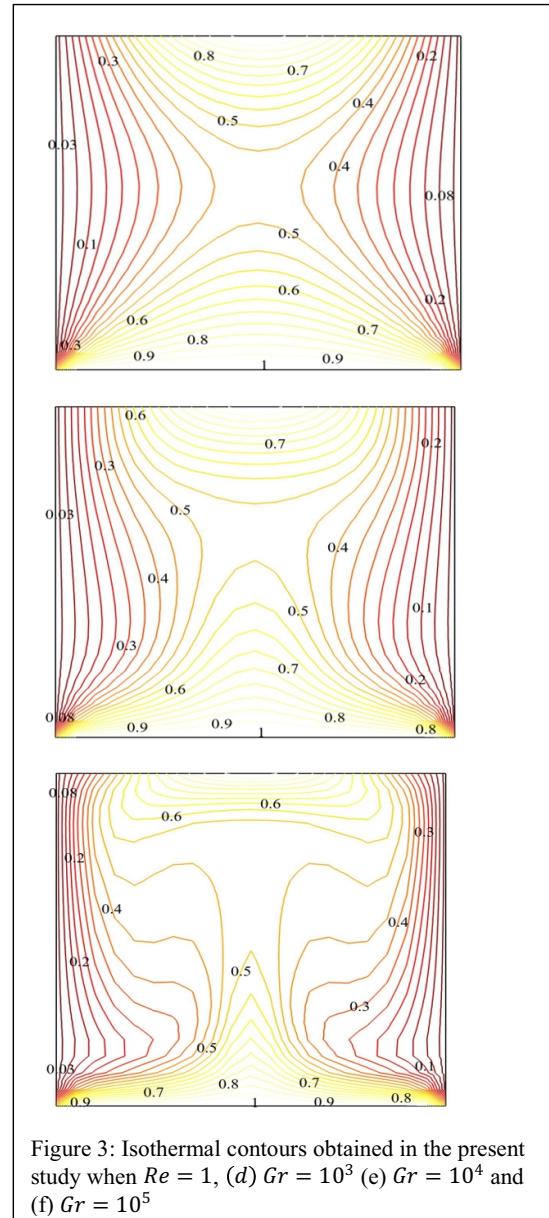
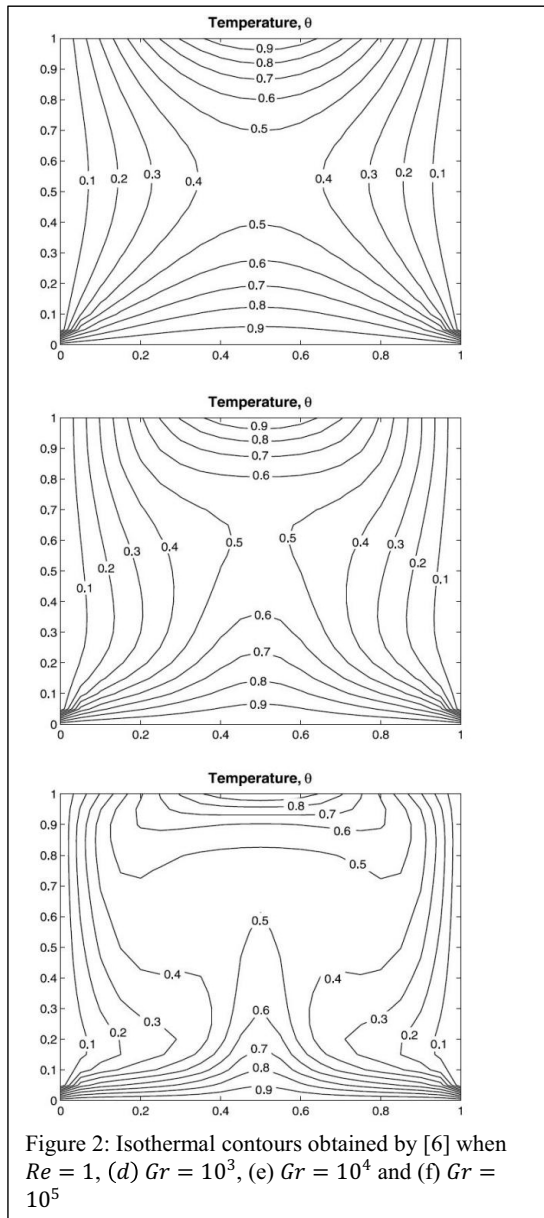
$$\overline{Nu} = \int_0^1 \frac{\partial \phi}{\partial X} dY \quad (7)$$

2.4 Computational procedure

The principal equations and the imposed boundary conditions were resolved using COMSOL Multiphysics 5.4 software package. The stability of the solution was ensured using the P2-P1 Lagrange elements and the Galerkin least square methods. The convergence criterion was set to 10^6 . The mesh of the geometry was generated using the free quadrilateral option. Grid sensitivity examination was done using COMSOL default settings. The findings were validated by comparing the values of Local Nusselt numbers for the cold left wall and heated bottom wall for $Re = 10$, and $Gr = 10^3$ and 10^5 . The results were unique and compare favorably with literature.

3. Results and Discussion

Figure 3 presents the comparison of isothermal contours obtained in the current work for $Gr = 10^3, 10^4$ and 10^5 corresponding respectively to isothermal contours from top to bottom for $Re = 1$ with those in equivalent adjacent positions in Figure 2 which were obtained under the same boundary conditions by Ducasse and Sibanda [6]. The comparison of both results shows good agreement.



Figures 4 and 5 show the plots of the velocity and isothermal contours when $Re = 100$, $Gr = 10^3$, 10^4 and 10^5 . In Figure 4, it was observed that when $Gr = 10^3$, a clockwise circulation was noticed in the square domain and when Gr was increased to 10^4 , close to the base of the cavity on the left in Figure 4, a small circulation in the counterclockwise direction was

observed. When Gr equals 10^5 , the size and intensity of the counterclockwise circulation were noticed to have increased. In Figure 5, the isothermal contours for $Gr = 10^3$ and $Gr = 10^4$ are similar in trend. The contours close to the cold vertical walls for both cases were seen to be nearly parallel near the vertical walls as expected and were distorted towards the center of the enclosure due to temperature increases. Furthermore, the hot isotherms were observed to be skewed leftwards close to the base and the top right side of the enclosure. Further increase of Gr from 10^4 to 10^5 resulted in symmetric contours around the hot bottom horizontal wall while the isotherms around the top horizontal wall were noticed to have further moved towards the right hand corner of the enclosure in comparison to the cases of $Gr = 10^3$ and 10^4 .

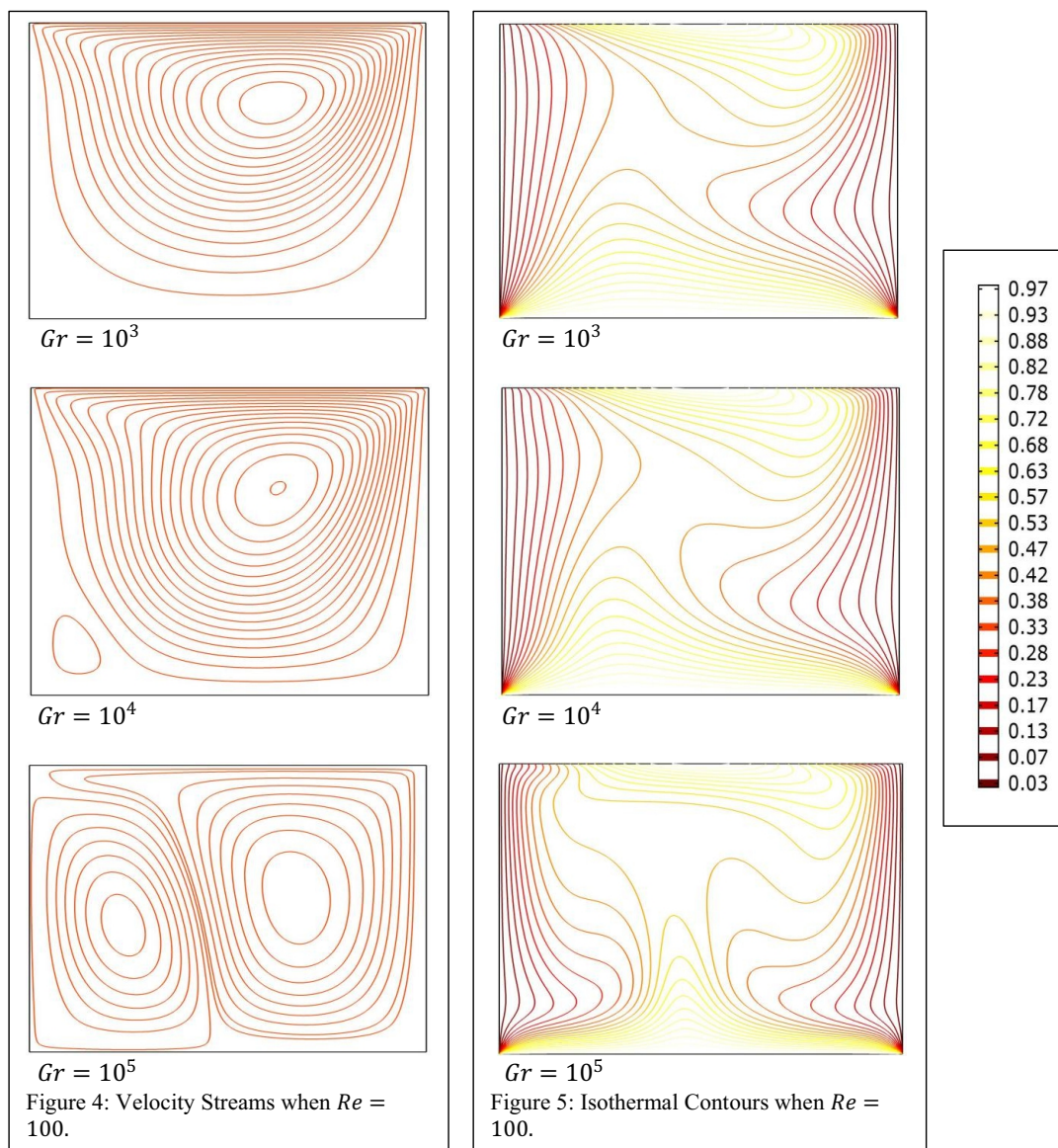
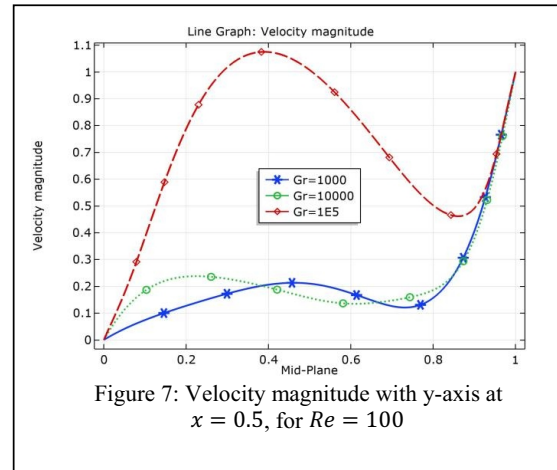
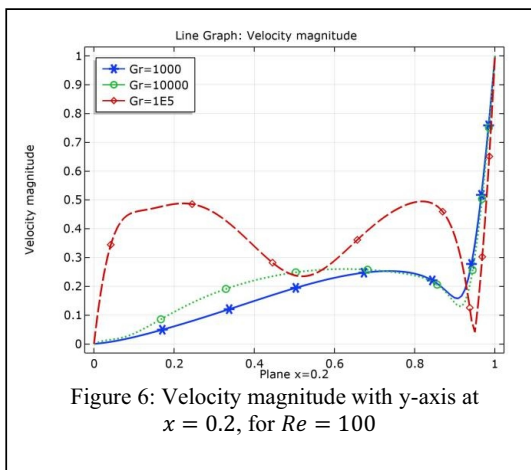


Figure 6 presents the plots of velocity magnitudes along the plane $x = 0.2$ and for $Re = 100$. When $Gr = 10^3$ and $Gr = 10^4$, the velocity magnitudes increased along the plane, each of the plots attained a local maximum and minimum, and then kept increasing until a maximum value at the top end of the plane due to temperature variation along the plane. For $Gr = 10^5$, the plot assumed a sinusoidal nature with the occurrence of local maximum and minimum points and finally attained its highest value of velocity at the top end of the plane. Similarly, Figure 7 is the velocity magnitude plots for plane $x = 0.5$ and $Re = 100$. It illustrates that at $Gr = 10^5$, in the interval of about $0.1 \leq Y \leq 0.8$, the mid-plane velocity is significantly greater than those of $Gr = 10^3$ and $Gr = 10^4$. Furthermore, at about the range of $0.9 \leq Y \leq 1$, the velocity magnitudes of all the plots assume the same value.



Figures 8 to 11 present the variation of local Nusselt number (Nu) along the left, right, top, and bottom walls of the square enclosure, respectively. In Figure 8, the plots for $Gr = 10^3$ and $Gr = 10^4$ were observed to almost superimpose each other throughout the span of the left wall. Furthermore, the local Nusselt number for both plots decreases along the length of the left cold wall until the upper end of the wall when the value of the local Nusselt number increased marginally. For $Gr = 10^5$, the local Nusselt number also decreases along the left side wall and then increased in the interval of $0.4 \leq Y \leq 0.9$ and finally decreases thereafter.

In Figure 9, for all the Gr values considered and in range of $0.1 \leq Y \leq 0.8$, the local Nusselt number decreased for all the plots and then increased throughout the interval of $0.8 \leq Y \leq 1.0$. The plots of Figure 10 assumed a sinusoidal nature which is in accordance with the imposed boundary condition on the top wall. In the first half-circle of the waveform, the plot for $Gr = 10^3$ and 10^4 attained the highest value of Nu while the converse was observed for the second half-circle while the trend of the plot for $Gr = 10^5$ was observed to be a reversal of the plots

for $Gr = 10^3$ and 10^4 . In Figure 11, in the interval $0 \leq X \leq 0.6$, the local Nusselt number decreased and then increased in the limit of $0.6 \leq X \leq 1.0$.

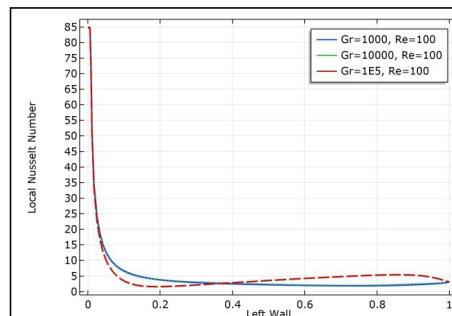


Figure 8: Variation of the local Nusselt number on the Left Wall

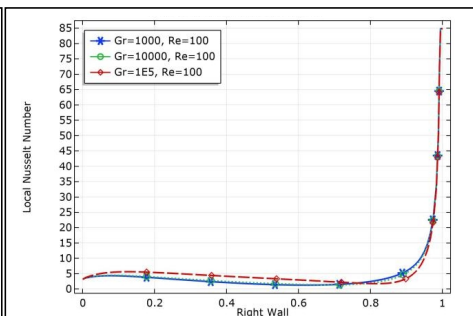


Figure 9: Variation of the local Nusselt number on the Right Wall

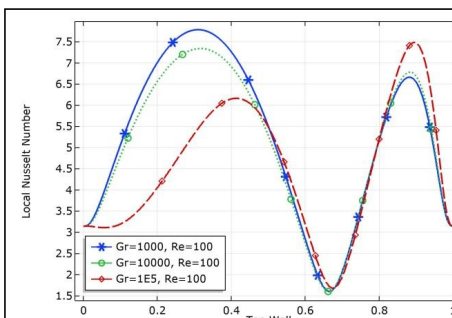


Figure 10: Variation of the local Nusselt number on the top wall

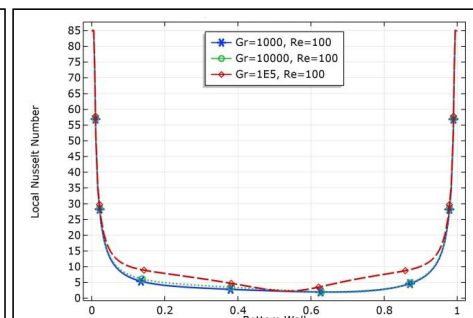


Figure 11: Variation of the local Nusselt number on the bottom wall

Figure 12 presents the relationship of the average Nusselt number on the walls of the enclosure with various Grashof numbers. The figures show that the highest average of Nusselt number occurred at the bottom wall of the cavity. Further examination revealed that left and right walls show comparable rates of heat transfer. In the limit $0 \leq Gr \leq 10^4$, the pattern of heat transferred by left, right, and top walls are similar; beyond this limit, the lowest rate of heat transfer occurred at the top wall.

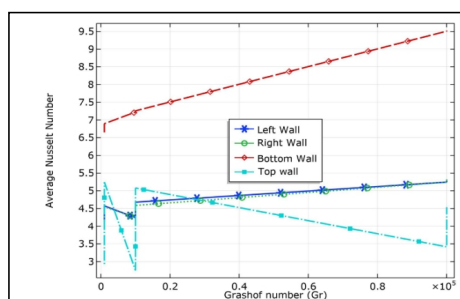


Figure 12: Variation of the average Nusselt number with Grashof number

Nomenclature

Gr	Grashof number
L	The dimensionless Length of the square cavity
Nu	local Nusselt number
$\overline{Nu}$	Average Nusselt number
P	dimensionless pressure
Pr	Prandtl number
Re	Reynolds number
U, V	dimensionless velocity components
X, Y	dimensionless Cartesian coordinates
φ	dimensionless temperature

4. Conclusion

The present investigation examined the implications of Reynolds and Grashof on the rate of local and average heat transfer, and the fluid flow characteristics inside an enclosure whose walls were subjected to various heating conditions. The findings revealed that the highest rate of heat transfer occurred near the bottom wall of the enclosure while the highest local Nusselt numbers on the vertical walls were found to have maximum numbers near the bottom corner and the top corner of the walls. The bottom wall, however, was noticed to have the highest local Nusselt number at each of its ends. The mid-plane velocity in the square enclosure was found to be highest at $Y = 0.4$.

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